DEHUMIDIFICATION IMPROVEMENT USING HEAT PIPES IN AIR CONDITIONING SYSTEMS

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***Abstract :** Heat pipes were first introduced by NASA for The Apollo Space program. The main factors that has led to the rise in the adaptation of heat pipes as heat exchangers are excellent thermal conductivities and no requirement of additional power for their operation. Using copper-based heat pipes, effective thermal conductivities of 1000 to 10,000 W/m-K have been achieved. This project aims to incorporate heat pipes into modern day HVAC systems in order to improve the performance parameters like humidity removal, cooling, tonnage required and hence, provide a better air conditioning system solution for both household and industrial HVAC applications. The paper consists of a comparison between a conventional air conditioning system and a system with heat pipes on the basis of humidity removal capacity at colder temperatures, along with an insight to the analysis of refrigerants which can possibly be used in such systems. A study has also been done in order to determine the compatibility of incorporation of heat pipes into existing systems.*

Keywords - Air Conditioning, Dehumidification, Evaporator, Heat Pipes, Psychometry, Refrigerants

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Nomenclature:
A: Surface area of heat exchange in the evaporator
C_f: Correction Factor
$C_{p,m}$: Specific heat of moist air
D: Diameter of evaporator coils
H: Height of the evaporator
HX: Heat Exchanger
L: Length of the evaporator
LHL: Latent Heat Load
LMTD: Log Mean Temperature Difference
m_a: Mass flow rate of air
m_r: Mass flow rate of refrigerant
N: Number of passes of pipes in evaporator
R.E.: Refrigeration Effect
RSHF: Room Sensible Heat Factor
SHL: Sensible Heat Load
THL: Total Heat Load
$T_{c,i}$: Inlet temperature of refrigerant
$T_{c,o}$: Exit temperature of refrigerant
$T_{h,I}$: Denoted temperature of ambient air (conventional)
$T_{h,o}$: Denoted temperature of conditioned air in room (conventional)
$T_{h,i1}$: Denoted temperature of ambient air (using heat pipe based evaporator)
$T_{h,i2}$: Denoted temperature of air entering the evaporator after precooling (using heat pipe based evaporator)
$T_{h,02}$: Denoted temperature of air exiting the evaporator (using heat pipe based evaporator)
ω_i: Denoted humidity ratio of ambient air (conventional)
Δh: Enthalpy change of refrigerant inside the evaporator
ΔT_1: Temperature difference between two fluids on the hotter side
ΔT_2: Temperature difference between two fluids on the cooler side

1. Introduction

Most of the modern day air conditioning systems rely heavily on temperature reduction to increase their energy efficiency ratings[1]. The moisture removal is not paid enough attention and so, the latent heat removal capacity of such systems are compromised in colder temperatures. Humidity, though not being of primary importance in household systems, play a vital role in cold storage and food preservation industries, libraries and hospitals. The cooler regions are therefore in need of better humidity control systems. Researches were carried out extensively to solve this problem [3][1]. Application of heat pipes [4] showed promising results in the research conducted worldwide, doing the intended job in an efficient way [1]. So, an evaporator unit upgraded with a heat pipe assembly can solve this problem, since heat pipes need no external power source to operate.

2. Objective

The objective of this project is to come up with a design of a heat pipe-incorporated evaporator, which could be used even in the existing systems. The moisture removal capacity and latent heat loads of both an existing conventional air conditioning system and the designed heat pipe incorporated system were calculated and compared, with a series of the settings. In order to show that heat pipe-based evaporator can solve the problem of low moisture removal, two ambient conditions were considered with different temperatures and humidity content of air, and the performance of both the systems were determined using calculations. A study on the refrigerants that can be used in the evaporator was done to determine the best refrigerants on the basis of the work required for compression. Some of the most commonly used freons, as well as newer ones like ammonia and carbon dioxide were being compared to show contrast in operations. Measured values were compared with actual values in order to show that heat pipes could be infused with the already existing systems.

3. Design

3.1 Conventional Air Conditioners without heat pipes

To determine the performance of conventional system, a 1-ton unit, with 100% outside air was studied. The bypass factor was kept zero for an ideal case comparison between the two systems. Unit mass flow rate of air was taken. Two cases of ambient were:

1. $T_{h,i}$=32℃, ω_i=20g/ kg of dry air (Colder and drier condition)
2. $T_{h,i}$=38℃, ω_i=28g/ kg of dry air (Hotter and humid condition)

The air conditioning settings were taken from 18℃ to 22℃. The specific heat of moist air [5] was calculated using:

$$C_{p,m}=1.005+1.82\omega_{i'} \qquad (3.1)$$

Using this, the sensible load was calculated as follows:

$$SHL=m_aC_{p,m}(T_{h,i}- T_{h,o}) \qquad (3.2)$$

The humidity of cooled air, ω_0, was taken from the psychometric chart[i], and using that, the latent load was calculated:

$$LHL=h_{T,\omega}(\omega_{i-}\ \omega_0) \qquad (3.3)$$

Total load came out as,

$$THL=SHL+LHL \qquad (3.4)$$

As a performance parameter, Room Sensible Heat Factor was calculated as:

$$RSHF=SHL/(SHL+LHL) \qquad (3.5)$$

3.2 Using evaporator unit with heat pipes

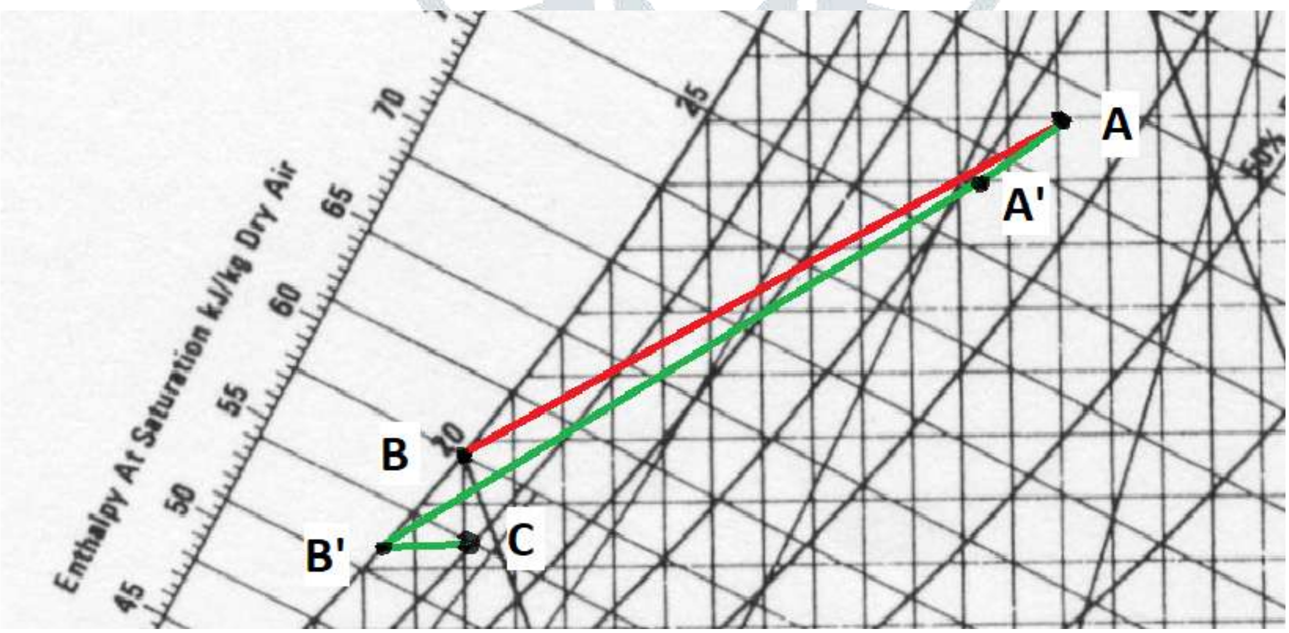
The value of SHL remained same as the temperature drop required was kept the same as before. The value of ambient humidity is not the humidity entering the evaporator and is denoted by ω_{i1}. However, the air was precooled using heat pipes before entering the evaporator. 1kW of heat was made to be removed by heat pipes [6], and the temperature before entering evaporator was calculated as:

$$1kW=m_aC_{p,m}(T_{h,i1}- T_{h,i2}) \qquad (3.6)$$

The value of SHL on evaporator remained same as the temperature drop required was kept the same as before. So, temperature at the other side of evaporator, $T_{h,02}$, will be:

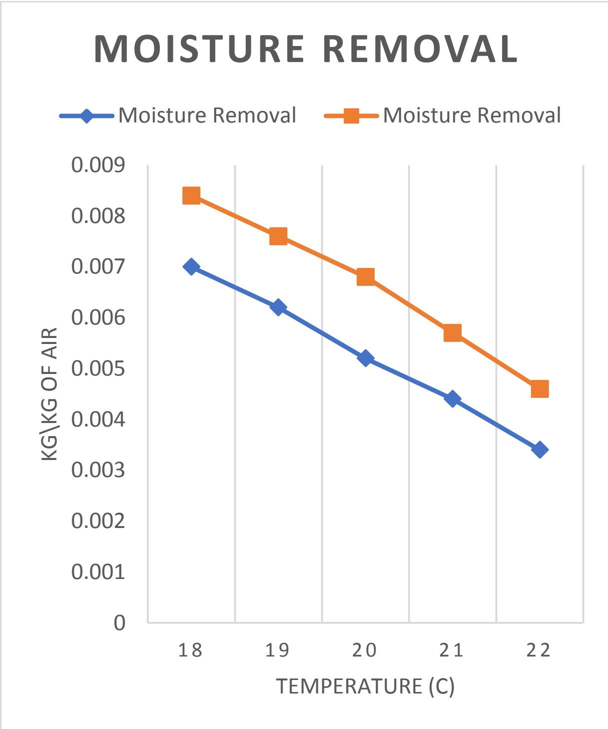
$$T_{h,i2}- T_{h,02=}\ T_{h,i}- T_{h,o} \qquad (3.7)$$

Hence, using these temperatures, humidity was evaluated using psychometric chart as ω_{i2} at temperature $T_{h,i2}$ and ω_{02} at temperature $T_{h,02}$. The values of LHL and RSHF were calculated in the same previous way at the five settings and both the ambient conditions.



3.3 Result of humidity removal comparison

The data obtained from both the cases were collected and tabulated to obtain a graphical representation of the results.



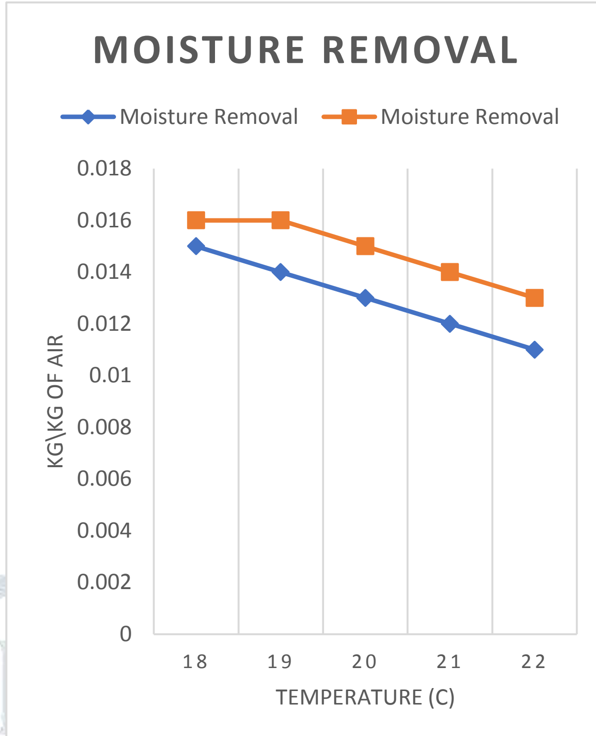


Fig.3.2: Comparison of humidity removal between air conditioning system at (i) 32℃ (ii) 38℃

The increase in latent heat removal capacity of the two systems was calculated by subtracting the latent heat loads of the two systems. Hence, a graph was plotted showing the increase in refrigeration effect of the two systems at various setting temperatures.

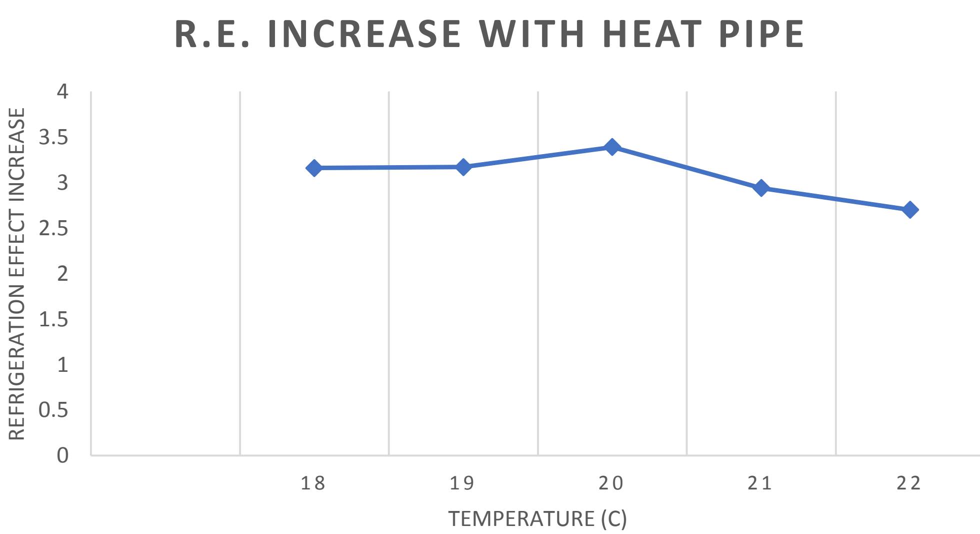


Fig.3.3: Relationship between the increase in refrigeration effect and the setting temperature

These charts gave the following results.

1) The moisture removal capacity of heat pipe-based evaporators is not only more than than their conventional counterparts, but also, this difference is significant at lower ambient temperatures.

2) The increase in moisture removal capacity and the refrigeration increase was most significant at the setting of 20^0C. Hence, the comparison of the refrigerants was done at this optimal setting.

4. Refrigerant Analysis

Several refrigerants were compared on the basis of the work required to compress them. The setting temperature was 20^0C. The enthalpy was calculated from Mollier chart [7] of that refrigerant at the recommended suction and discharge pressures [8][9](iii). The refrigeration effects were compared in both conventional and heat pipe incorporated systems, and hence, the variation in the compressor work was tabulated for each refrigerant. Taking mass flow rate of refrigerant as 0.8kg/s, refrigerant enthalpy change in both cases was determined by:

$$THL = m_r(\Delta h) \qquad (4.1)$$

Table 4.1: Work comparison between refrigerants

Refrigerant	Work required (kJ)	Suction Pressure (bar)
R-22	8	4.08
R-134a	6.8	2.58
R 717(Ammonia)	32	2.37
R 744(CO_2)	7.4	26.18
R 507	5	2.5
R 410a	16	1.632
R407c	9	3.4
R 404a	5	1.632

5. Incorporation into Current Systems

In order test for the compatibility with current systems, the surface area of heat exchange in evaporator tubes were compared for a 1-ton AC. Measured value of the surface area was compared to the calculated surface required in case of heat pipe-based evaporator. LG L Prima was measured for the dimensions of the evaporator. Dimensions D, L and H was noted. Number of passes in evaporator was calculated as:

$$N=(H/D) \qquad (5.1)$$

Hence, the area of heat exchange came out to be:

$$A=\pi DLN \qquad (5.2)$$

This came out to be 0.51 sq. meters.

To estimate the surface area required, LMTD method [ii] was used for an ambient of 32℃ and temperature setting of 20℃ for counter flow HX. A correction factor [11] was used for cross flow case. The temperature difference at the two ends were:

$$\Delta T_1= T_{h,i2}- T_{c,o}\ ,\ \ \Delta T_2= T_{h,02}- T_{c,I} \qquad (5.3)$$

$$LMTD=(\Delta T_1-\Delta T_2)/\ln(\Delta T_1/\Delta T_2) \qquad (5.4)$$

For 1-Ton, Q=3.51685kW. Using overall heat transfer equation:

$$Q=AU(LMTD)C_f \qquad (5.5)$$

Hence, the area came out to be 0.47 sq. meters.
This result is close to the measured area.

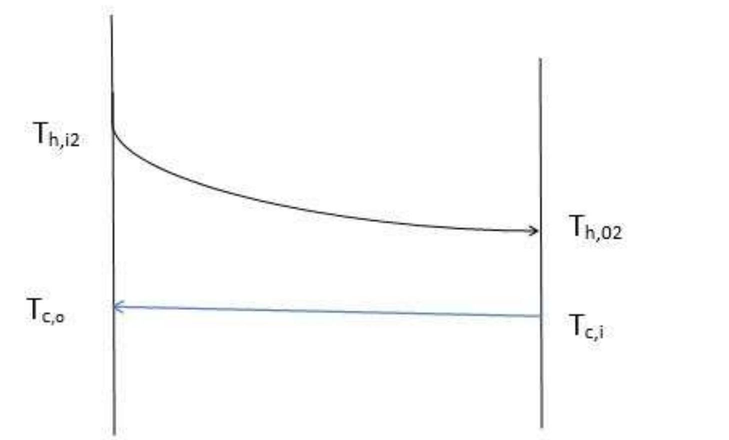


Fig.5.1: LMTD diagram of evaporator assembly with heat pipe and R-22 as cold fluid

6. CONCLUSION

The project laid emphasis on the functioning benefits of heat pipe incorporation in evaporator and showed that the both the moisture removal and hence the refrigeration effect increased even when the ambient conditions were colder and drier. Increase of an average of 2 grams per kg of dry air per second of humidity removal and 3 kJ of Refrigeration Effect increase were noted. The heat pipe-based system showed better functioning at lower ambient. Also, since the heat pipes require no external power source, the effect can be achieved at lower power and tonnage than the conventional systems. Though this setup requires an initial capital, cost estimations and performance reviews worldwide [12] proved that the capital recovery duration is not only rather short (1-1.5 years) but also in long term saves energy and maintenance costs. Refrigerant analysis shows carbon dioxide as a viable refrigerant for ACs as freons are facing environmental regulations. The study of area calculation proved that not only is this system more efficient, but also that the already existing systems can be upgraded to work like this.

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